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Ready Reference

FOR

ENGINEERS AND STEAM USERS,

BY

JAMES B. STANWOOD, M. E.

PUBLISHED BY

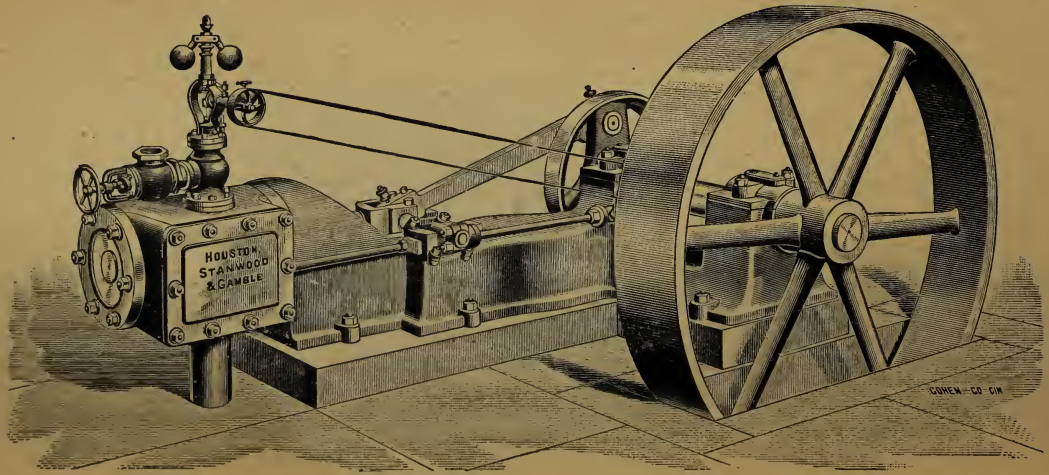
HOUSTON, STANWOOD & GAMBLE,

CINCINNATI, O.

1895.

PRESS OF GEO. P. HOUSTON; CINCINNATI.

PRICE, 25 CENTS.



The Houston, Stanwood & Gamble Standard Simple Engine.

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JAMES B. STANWOOD, M. E.

Member of American Society of Mechanical Engineers,
Director of Cincinnati Technical School.

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1895.

SECOND EDITION.

PRICE, TWENTY-FIVE CENTS.

PREFACE.

THIS partial collection of data and experience has been the result of seventeen years spent in the service of Steam Engineering. If this little work can be of value to others, its object will be accomplished.

J. B. S.

August, 1892.

THE STEAM ENGINE.

WORK, consists of the sustained exertion of force through space. The *unit* of work, the *foot-pound*, is a force of 1 lb., exerted through 1 ft. space.

The work done in lifting 1 pound 10 feet, or 10 pounds 1 foot is 10 foot-pounds.

POWER, is the rate of work, or, the number of foot-pounds exerted in a unit of time. The *unit* of power is the *horse-power*, and equals 33,000 foot-pounds exerted in a minute, or 550 foot-pounds exerted in a second, or 1,980,000 foot-pounds exerted in an hour.

An engine developing 50 horse-power, exerts 27,500 foot-pounds per second, 1,650,000 foot-pounds in a minute. It could raise (friction neglected) 41,250 pounds, 40 feet in one minute.

A belt running over a pulley at 4,000 feet per minute, pulling with a force of 240 pounds, (fair load for a 4-inch belt) will transmit

$$\frac{240 \times 4{,}000}{33{,}000} = 30 \text{ horse-power (nearly).}$$

If moving at 1,100 feet per minute, the result would be

$$\frac{240 \times 1{,}100}{33{,}000} = 8 \text{ horse-power.}$$

A gear-wheel, the cogs of which transmit a pressure of 1,800 pounds (fair load for 1½" pitch 6" face) to the cogs of its mate, the periphery velocity of the wheels being 10 feet per second, transmits

$$\frac{1{,}800 \times 10}{550} = 33 \text{ horse-power.}$$

If speed was 360 feet per minute, it would transmit

$$\frac{1{,}800 \times 360}{33{,}000} = 20 \text{ horse-power.}$$

The HORSE-POWER developed by a STEAM ENGINE consists of two primary factors, *piston speed*, and *total average pressure* of steam upon the piston.

PISTON SPEED depends upon the stroke of engine and the number of revolutions per minute.

An engine with stroke of 12 inches, making 300 revolutions per minute, has a piston speed of

$$\frac{2 \times 12 \times 300}{12} = 600 \text{ feet per minute.}$$

Piston speed of an engine with 24" stroke at 150 revolutions per minute:

$$\frac{2 \times 24 \times 150}{12} = 600 \text{ feet per minute.}$$

TOTAL AVERAGE PRESSURE depends on *area of piston* and *mean effective pressure* per square inch exerted on piston throughout stroke. The mean effective pressure (M. E. P.) in any case can only be accurately obtained by means of the steam engine indicator, and depends upon the load engine is carrying.

The AREA OF PISTON in square inches can be found in Table IV.

The maximum MEAN EFFECTIVE PRESSURE, per square inch, that can be obtained by automatic cut-off engines with varying cut-offs, is shown by Table I.

TABLE I.

MEAN EFFECTIVE PRESSURES FOR VARYING CUT-OFFS AND INITIAL STEAM PRESSURES; NON-CONDENSING ENGINES.

Initial Pressure.	1/10 Cut-off.	1.5/100 Cut-off.	1/5 Cut-off.	1/4 Cut-off.	3/10 Cut-off.	3.5/100 Cut-off.	4/10 Cut-off.	1/2 Cut-off.
40	3.65	9.05	13.46	17.34	20.75	23.70	26.22	30.50
45	5.42	11.32	16.15	20.39	24.13	27.32	30.08	34.75
50	7.19	13.59	18.85	23.45	27.50	30.94	33.95	39.00
55	8.96	15.86	21.54	26.50	30.87	34.56	37.81	43.25
60	10.73	18.12	24.24	29.56	34.24	38.18	41.68	47.50
65	12.50	20.39	26.93	32.61	37.61	41.80	45.54	51.75
70	14.27	22.66	29.63	35.67	40.98	45.42	49.41	56.00
75	16.04	24.92	32.32	38.72	44.35	49.05	53.27	60.25
80	17.81	27.19	35.02	41.78	47.72	52.68	57.14	64.50
85	19.58	29.46	37.71	44.83	51.09	56.31	61.00	68.75
90	21.36	31.72	40.41	47.89	54.46	59.94	64.87	73.00
95	23.13	33.93	43.10	50.94	57.83	63.57	68.73	77.25
100	24.90	36.26	45.80	54.00	61.20	67.20	72.60	81.50

With throttling engines results are obtained dependent upon proportion of steam-ports, travel of valve, piston speed, size of governor, and design of engine.

With Houston, Stanwood & Gamble's Standard Slide-Valve Engines, in which the steam-ports are 9 per cent. of the

piston area, valve-travel nearly four times the width of steam-ports, piston speed 450 feet per minute, and governors ¼ size of piston diameter, Table II., column 2, gives nearly the maximum M. E. P. for varying pressure; column 3 gives same with governors one size larger.

To FIND HORSE-POWER of an engine, multiply the piston speed in feet per minute by the area of piston in square inches, divide by 33,000. This gives horse-power for one pound mean effective pressure. Determine M. E. P. from indicator diagram, or select same from Tables I. or II., and multiply product found by this amount.

A Houston, Stanwood & Gamble Slide-Valve Engine, 13" piston, 20 inch stroke, with standard size governor, makes 140 revolutions per minute at 60 pounds boiler pressure, what maximum horse-power will it develop at that speed?

$$\text{Piston Speed} = \frac{2 \times 20 \times 140}{12} = 466\tfrac{2}{3} \text{ feet per minute.}$$

Area 13 inch Piston = 132.7 sq. inch. (See Table IV.)

$$\frac{466\tfrac{2}{3} \times 132.7}{33{,}000} = 1.9 \text{ horse-power per each pound of M. E. P.}$$

M. E. P. for 60 pounds = 30. (See Table II.)

$1.9 \times 30 = 57$ = horse-power.

HEAT AND STEAM.—A thermal unit is the amount of heat necessary to raise the temperature of one pound of water one degree Fahrenheit.

To boil water heat must be applied to raise its temperature to the boiling point (which varies with the pressure that exists above its surface); and to this enough more heat (latent heat) must be added to convert the water into steam.

EXAMPLE.—To raise one pound of water from 60° Fahr. and convert it into steam at 212° requires

212° — 60° =	152 thermal units.
latent heat	965 thermal units. (See table III.)
total heat	1117 thermal units (nearly).

EXAMPLE.—To raise one pound of water from 100° Fahr. and convert it into steam at 75 pounds pressure, requires (nearly).

320° — 100° =	220 thermal units.
latent heat	888 thermal units. (See table III.)
total heat	1108 thermal units (nearly).

NOTE.—320° is boiling point of water under 75 pounds pressure by gauge, or 90 lbs. above a perfect vacuum. 888 thermal units are required to transform one pound of water into steam at that temperature.

The total heat as above calculated is not strictly correct, as a small allowance should be made for the specific heat of water. The above illustration embodies the principle.

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

TABLE II.

SAFE MAXIMUM MEAN EFFECTIVE PRESSURES FOR HOUSTON, STANWOOD & GAMBLE'S STANDARD SLIDE-VALVE ENGINES, (NON-CONDENSING.) PISTON SPEED, 450 FEET.

Pressure in Steam Pipe at Governor. LBS.	Governor. ¼ Piston Size. LBS.	Governor. ¼ Piston Size. + ½-inch. LBS.
40	19.2	22.4
45	22.2	25.9
50	24.6	28.7
55	27.6	32.2
60	30.	35.
65	33.	38.5
70	36.	42.0
75	38.4	45.8
80	41.4	48.3
85	43.8	51.1
90	46.8	54.6
95	49.8	58.1
100	52.2	60.9

It is not safe to estimate on more than 80 pounds pressure for continuous duty, as in a few special cases only is steam pressure kept constantly above 80 pounds.

THE PROPERTIES OF STEAM.

Table III. gives the properties of steam, deduced from *D. K. Clark's Manual.*

Column 1—P, varying pressures from 1 pound above a perfect vacuum to 165 pounds.

Column 2—T, temperatures corresponding to pressures P.

Column 3—U, total units of heat necessary to convert 1 pound of water at 32° Fahr. into 1 pound of steam corresponding to pressure P.

Column 4—L, latent heat, in units of heat.

Column 5—W, weight of a cubic foot of steam at pressures P.

Column 6—A value, K, 13,750 times W, by which the steam in pounds per hour per indicated horse-power can be computed.

TABLE III.

PROPERTIES OF SATURATED STEAM.

(Compiled from D. K. Clark's Manual.)

Total pressure per square in.	Temperature in Fahrenheit degrees	Total heat, in Fahr. degrees, from water at 32° F.	Latent heat, Fahrenheit degrees	Weight of one cubic foot.	13,750 times the weight of one cubic foot.
Lbs.	Fahr.	Fahr.	Fahr.	Lbs.	Lbs.
1	102.1	1112.5	1042.9	.003	41.25
2	126.3	1119.7	1025.8	.0058	79.75
3	141.6	1124.6	1015.0	.0085	116.875
4	153.1	1128.1	1006.8	.0112	154.00
5	162.3	1130.9	1000.3	.0138	189.75
6	170.2	1133.3	994.7	.0163	224.125
7	176.9	1135.3	990.0	.0189	259.875
8	182.9	1137.2	985.7	.0214	294.25
9	188.3	1138.8	981.9	.0239	328.625
10	193.3	1140.3	978.4	.0264	363.00
11	197.8	1141.7	975.2	.0289	397.375
12	202.0	1143.0	972.2	.0314	431.750
13	205.9	1144.2	969.4	.0338	464.75
14	209.6	1145.3	966.8	.0362	497.75
14 7	212.0	1146.1	965.2	.0380	522.500
15	213.1	1146.4	964.3	.0387	532.125
16	216.3	1147.4	962.1	.0411	565.125
17	219.6	1148.3	959.8	.0435	598.125
18	222.4	1149.2	957.7	.0459	631.125
19	225.3	1150.1	955.7	.0483	664.125
20	228.0	1150.9	952.8	.0507	697.125
21	230.6	1151.7	951.3	.0531	730.125
22	233.1	1152.5	949.9	.0555	763.125
23	235.5	1153.2	948.5	.0580	797.500
24	237.8	1153.9	946.9	.0601	826.375
25	240.1	1154.6	945.3	.0625	859.375
26	242.3	1155.3	943.7	.0650	893.750
27	244.4	1155.8	942.2	.0673	925.375
28	246.4	1156.4	940.8	.0696	957.00
29	248.4	1157.1	939.4	.0719	988.625
30	250.4	1157.8	937.9	.0733	1007.875
31	252.2	1158.4	936.7	.0766	1053.250
32	254.1	1158.9	935.3	.0789	1084.875
33	255.9	1159.5	934.0	.0812	1116.50
34	257.6	1160.0	932.8	.0835	1148.125
35	259.3	1160.5	931.6	.0858	1179.75
36	260.9	1161.0	930.5	.0881	1211.375
37	262.6	1161.5	929.3	.0905	1244.375
38	264.2	1162.0	928.2	.0929	1277.375
39	265.8	1162.5	927.1	.0952	1309.00
40	267.3	1162.9	926.0	.0974	1339.25
41	268.7	1163.4	924.9	.0996	1369.50
42	270.2	1163.8	923.9	.1020	1402.50
43	271.6	1164.2	922.9	.1042	1432.75
44	273.0	1164.6	921.9	.1065	1464.375
45	274.4	1165.1	920.9	.1089	1497.375
46	275.8	1165.5	919.9	.1111	1527.625
47	277.1	1165.9	919.0	.1133	1557.875
48	278.4	1166.3	918.1	.1156	1589.500
49	279.7	1166.7	917.2	.1179	1621.125

TABLE III.—Continued.

Total pressure per square in.	Temperature in Fahrenheit degrees	Total heat, in Fahr. degrees, from water at 32° F.	Latent heat, Fahrenheit degrees	Weight of one cubic foot.	13,750 times the weight of one cubic foot.
Lbs.	Fahr.	Fahr.	Fahr.	Lbs.	Lbs.
50	281.0	1167.1	916.3	.1202	1652.75
51	282.3	1167.5	915.4	.1224	1683.00
52	283.5	1167.9	914.5	.1246	1713.25
53	284.7	1168.3	913.6	.1269	1744.875
54	285.9	1168.6	912.8	.1291	1775.125
55	287.1	1169.0	912.0	.1314	1806.75
56	288.2	1169.3	911.2	.1336	1837.00
57	289.3	1169.7	910.4	.1364	1875.50
58	290.4	1170.0	909.6	.1380	1897.50
59	291.6	1170.4	908.8	.1403	1929.125
60	292.7	1170.7	908.0	.1425	1959.375
61	293.8	1171.1	907.2	.1447	1989.625
62	294.8	1171.4	906.4	.1469	2019.875
63	295.9	1171.7	905.6	.1493	2052.875
64	296.9	1172.0	904.9	.1516	2084.5
65	298.0	1172.3	904.2	.1538	2114.75
66	299.0	1172.6	903.5	.1560	2145.00
67	300.0	1172.9	902.8	.1583	2176.625
68	300.9	1173.2	902.1	.1605	2206.875
69	301.9	1173.5	901.4	.1627	2237.125
70	302.9	1173.8	900.8	.1648	2266.00
71	303.9	1174.1	900.3	.1670	2296.25
72	304.8	1174.3	899.6	.1692	2326.50
73	305.7	1174.6	898.9	.1714	2356.75
74	306.6	1174.9	898.2	.1736	2387.00
75	307.5	1175.2	897.5	.1759	2418.625
76	308.4	1175.4	896.8	.1782	2450.25
77	309.3	1175.7	896.1	.1804	2480.50
78	310.2	1176.0	895.5	.1826	2510.75
79	311.1	1176.3	894.9	.1848	2541.00
80	312.0	1176.5	894.3	.1869	2569.875
81	312.8	1176.8	893.7	.1891	2600.125
82	313.6	1177.1	893.1	.1913	2630.375
83	314.5	1177.4	892.5	.1935	2660.625
84	315.3	1177.6	892.0	.1957	2690.875
85	316.1	1177.9	891.4	.1980	2722.50
86	316.9	1178.1	890.8	.2002	2752.75
87	317.8	1178.4	890.2	.2024	2783.00
88	318.6	1178.6	889.6	.2044	2810.50
89	319.4	1178.9	889.0	.2067	2842.125
90	320.2	1179.1	888.5	.2089	2872.375
91	321.0	1179.3	887.9	.2111	2902.625
92	321.7	1179.5	887.3	.2133	2932.875
93	322.5	1179.8	886.8	.2155	2963.125
94	323.3	1180.0	886.3	.2176	2992.00
95	324.1	1180.3	885.8	.2198	3022.25
96	324.8	1180.5	885.2	.2219	3051.125
97	325.6	1180.8	884.6	.2241	3081.375
98	326.3	1181.0	884.1	.2263	3111.625
99	327.1	1181.2	883.6	.2285	3141.875
100	327.9	1181.4	883.1	.2307	3172.125
101	328.5	1181.6	882.6	.2329	3202.375

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Mem. Am. Phil. Soc.
Lancaster, Pa.

Stanwood, James B

A ready reference for engineers and steam users. Cincinnati, Houston Stanwood & Gamble, 1895.

Total pressure per square in.	Temperature in Fahrenheit degrees	Total heat, in Fahr. degrees, from water at 32° F.	Latent heat, Fahrenheit degrees	Weight of one cubic foot.	13,750 times the weight of one cubic foot.
Lbs.	Fahr.	Fahr.	Fahr.	Lbs.	Lbs.
102	329.1	1181.8	882.1	.2351	3232.625
103	329.9	1182.0	881.6	.2373	3262.875
104	330.6	1182.2	881.1	.2393	3290.375
105	331.3	1182.4	880.7	.2414	3319.25
106	331.9	1182.6	880.2	.2435	3348.125
107	332.6	1182.8	879.7	.2456	3377.00
108	333.3	1183.0	879.2	.2477	3405.875
109	334.0	1183.3	878.7	.2499	3436.125
110	334.6	1183.5	878.3	.2521	3466 375
111	335.3	1183.7	877.8	.2543	3496.625
112	336.0	1183.9	877.3	.2564	3525.50
113	336.7	1184.1	876.8	.2586	3555.75
114	337.4	1184.3	876.3	.2607	3584.625
115	338.0	1184.5	875.9	.2628	3613.5
116	338.6	1184.7	875.5	.2649	3642.375
117	339.3	1184.9	875.0	.2652	3646.5
118	339.9	1185.1	874.5	.2674	3676.75
119	340.5	1185.3	874.1	.2696	3707.00
120	341.1	1185.4	873.7	.2738	3764.75
121	341.8	1185.6	873.2	.2759	3793.625
122	342.4	1185.8	872.8	.2780	3822.5
123	343.0	1186.0	872.3	.2801	3851.375
124	343.6	1186.2	871.9	.2822	3880.25
125	344.2	1186.4	871.5	.2845	3911.875
126	344.8	1186.6	871.1	.2867	3942.125
127	345.4	1186.8	870.7	.2889	3972.375
128	346.0	1186.9	870.2	.2911	4002.625
129	346.6	1187.1	869.8	.2933	4032.875
130	347.2	1187.3	869.4	.2955	4063.125
131	347.8	1187.5	869.0	.2977	4093.375
132	348.3	1187.6	868.6	.2999	4123.625
133	348.9	1187.8	868.2	.3020	4152.50
134	349.5	1188.0	867.8	.3040	4180.00
135	350.1	1188.2	867.4	.3060	4207.50
136	350.6	1188.3	867.0	.3080	4235.00
137	351.2	1188.5	866.6	.3101	4263.875
138	351.8	1188.7	866.2	.3121	4291.375
139	352.4	1188.9	865.8	.3142	4320.25
140	352.9	1189.0	865.4	.3162	4347.75
141	353.5	1189.2	865.0	.3184	4378.00
142	354.0	1189.4	864.6	.3206	4408.25
143	354.5	1189.6	864.2	.3228	4438.5
144	355.0	1189.7	863.9	.3250	4468.75
145	355.6	1189.9	863.5	.3273	4500.375
146	356.1	1190.0	863.1	.3294	4529.25
147	356.7	1190.2	862.7	.3315	4558.125
148	357.2	1190.3	862.3	.3336	4587.00
149	357.8	1190.5	861.9	.3357	4615.875
150	358.3	1190.7	861.5	.3377	4643.375
155	361.0	1191.5	859.7	.3484	4790.50
160	363.4	1192.2	857.9	.3590	4936.25
165	366.0	1192.9	856.2	.3695	5080.625

TO COMPUTE THE AMOUNT OF STEAM ACCOUNTED FOR BY STEAM ENGINE INDICATOR, IN LBS. PER HORSE-POWER PER HOUR, FROM INDICATOR DIAGRAM.

There are two methods: 1st. By measurement of amount of steam in cylinder at cut-off. 2nd. By measurement of amount of steam in cylinder at time valve opens for release. The latter gives nearer the actual results, but the former is the best measure of steam entering the cylinder as steam.

Take following measurements on any indicator diagram:

1st. The pressure at cut-off above a vacuum = X.
2nd. " " compression " " = Y.
3rd. " " release " " = Z.
4th. The proportion of stroke finished at cut-off = A.
5th. " " " " " release = R.
6th. " " " " unfinished at compression = C.

In table III., col.6 find K_A corresponding to X.
K_c " " Y.
K_R " " Z.

Determine the clearance, D, as a proportion of the entire piston displacement, and from diagram the mean effective pressure, m. e. p.

Then by first method, (measuring steam at cut-off).

$$S = \text{Steam per ind. h. p. per hr.} = \frac{(A+D)\ K_A - (C+D)\ K_c}{\text{m. e. p.}}$$

Then by second method, (measuring steam at release).

$$S = \frac{(R+D)\ K_R - (C+D)\ K_c}{\text{m. e. p.}}$$

When clearance can not easily be determined upon, a rough approximation is as follows, which has the advantage of simplicity,

$$S = \frac{R\ K_R}{\text{m. e. p.}}$$

EXAMPLE.—An indicator diagram gives the following values:

Pressure at cut-off above a vacuum = 88½ lbs. = X.
" " compression " " = 16.2 lbs. = Y.
" " release " " = 28.4 lbs. = Z.
Proportion of stroke finished at cut-off = .26 = A.
" " " " release = .95 = R.
" " unfinished at compression = .10 = C.

Clearance = 0.04 (by estimate.)
Mean effective pressure = 36 pounds.

$K_A = 2{,}827$ $K_c = 572$ $K_R = 969$

First method:

$$S = \frac{(.26+.04)\ K_A - (.1+.04)\ K_c}{36} = \frac{.3 \times 2827 - .14 \times 572}{36}$$

$$= \frac{848.1 - 80.1}{36} = \frac{768}{36} = 21\tfrac{1}{3} \text{ lbs.}$$

Second method:

$$S = \frac{(.95+.04)\ K_R - (.1+.04)\ K_c}{36} = \frac{.99 \times 969 - .14 \times 572}{36}$$

$$= \frac{959. - 80.1}{36} = \frac{879}{36} = 24.4 \text{ lbs}$$

Approximate method: $S = \frac{.95\ K_R}{36} = \frac{920}{36} = 25.5$ lbs.

TABLE IV.

AREAS OF CIRCLES HAVING DIAMETERS VARYING FROM 1 INCH TO 100 INCHES.

Diam. in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.
1	0.7854	3 7/8	11.793	8 1/2	56.745
1 1/16	0.8866	3 15/16	12.177	8 5/8	58.426
1 1/8	0.9940	4	12.566	8 3/4	60.132
1 3/16	1.1075	4 1/16	12.962	8 7/8	61.862
1 1/4	1.2271	4 1/8	13.364	9	63.617
1 5/16	1.3530	4 3/16	13.772	9 1/8	65.396
1 3/8	1.4848	4 1/4	14.186	9 1/4	67.200
1 7/16	1.6229	4 5/16	14.606	9 3/8	69.029
1 1/2	1.7671	4 3/8	15.033	9 1/2	70.882
1 9/16	1.9175	4 7/16	15.465	9 5/8	72.759
1 5/8	2.0739	4 1/2	15.904	9 3/4	74.662
1 11/16	2.2365	4 9/16	16.349	9 7/8	76.588
1 3/4	2.4052	4 5/8	16.800	10	78.540
1 13/16	2.5800	4 11/16	17.257	10 1/8	80.515
1 7/8	2.7611	4 3/4	17.720	10 1/4	82.516
1 15/16	2.9483	4 13/16	18.190	10 3/8	84.540
2	3.1416	4 7/8	18.665	10 1/2	86.590
2 1/16	3.3380	4 15/16	19.147	10 5/8	88.664
2 1/8	3.5465	5	19.635	10 3/4	90.762
2 3/16	3.7584	5 1/8	20.629	10 7/8	92.885
2 1/4	3.9760	5 1/4	21.647	11	95.033
2 5/16	4.2000	5 3/8	22.690	11 1/8	97.205
2 3/8	4.4302	5 1/2	23.758	11 1/4	99.402
2 7/16	4.6664	5 5/8	24.850	11 3/8	101.623
2 1/2	4.9087	5 3/4	25.967	11 1/2	103.869
2 9/16	5.1573	5 7/8	27.108	11 5/8	106.139
2 5/8	5.4119	6	28.274	11 3/4	108.434
2 11/16	5.6723	6 1/8	29.464	11 7/8	110.753
2 3/4	5.9395	6 1/4	30.679	12	113.097
2 13/16	6.2126	6 3/8	31.919	12 1/8	115.466
2 7/8	6.4918	6 1/2	33.183	12 1/4	117.859
2 15/16	6.7772	6 5/8	34.471	12 3/8	120.276
3	7.0686	6 3/4	35.784	12 1/2	122.718
3 1/16	7.3662	6 7/8	37.122	12 5/8	125.184
3 1/8	7.6699	7	38.484	12 3/4	127.676
3 3/16	7.9798	7 1/8	39.871	12 7/8	130.192
3 1/4	8.2957	7 1/4	41.282	13	132.732
3 5/16	8.6180	7 3/8	42.718	13 1/8	135.297
3 3/8	8.9462	7 1/2	44.178	13 1/4	137.886
3 7/16	9.2807	7 5/8	45.663	13 3/8	140.500
3 1/2	9.6211	7 3/4	47.173	13 1/2	143.139
3 9/16	9.9680	7 7/8	48.707	13 5/8	145.802
3 5/8	10.320	8	50.265	13 3/4	148.489
3 11/16	10.679	8 1/8	51.848	13 7/8	151.201
3 3/4	11.044	8 1/4	53.456	14	153.938
3 13/16	11.416	8 3/8	55.088	14 1/8	156.699

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD

TABLE IV.—Continued.

Diam in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.
14¼	159.485	20⅜	326.051	26½	551.547
14⅜	162.295	20½	330.064	26⅝	556.762
14½	165.130	20⅝	334.101	26¾	562.002
14⅝	167.989	20¾	338.163	26⅞	567.267
14¾	170.873	20⅞	342.250	27	572.556
14⅞	173.782	21	346.361	27⅛	577.870
15	176.715	21⅛	350.497	27¼	583.208
15⅛	179.672	21¼	354.657	27⅜	588.571
15¼	182.654	21⅜	358.841	27½	593.958
15⅜	185.661	21½	363.051	27⅝	599.370
15½	188.692	21⅝	367.284	27¾	604.807
15⅝	191.748	21¾	371.543	27⅞	610.268
15¾	194.828	21⅞	375.826	28	615.753
15⅞	197.933	22	380.133	28⅛	621.263
16	201.062	22⅛	384.465	28¼	626.798
16⅛	204.216	22¼	388.822	28⅜	632.357
16¼	207.394	22⅜	393.203	28½	637.941
16⅜	210.597	22½	397.608	28⅝	643.594
16½	213.825	22⅝	402.038	28¾	649.182
16⅝	217.077	22¾	406.493	28⅞	654.839
16¾	220.353	22⅞	410.972	29	660.521
16⅞	223.654	23	415.476	29⅛	666.227
17	226.980	23⅛	420.004	29¼	671.958
17⅛	230.330	23¼	424.557	29⅜	677.714
17¼	233.705	23⅜	429.135	29½	683.494
17⅜	237.104	23½	433.731	29⅝	689.298
17½	240.528	23⅝	438.363	29¾	695.128
17⅝	243.977	23¾	443.014	29⅞	700.981
17¾	247.450	23⅞	447.699	30	706.860
17⅞	250.947	24	452.390	30¼	718.690
18	254.469	24⅛	457.115	30½	730.618
18⅛	258.016	24¼	461.864	30¾	742.644
18¼	261.587	24⅜	466.638	31	754.769
18⅜	265.182	24½	471.436	31¼	766.992
18½	268.803	24⅝	476.259	31½	779.313
18⅝	272.447	24¾	481.106	31¾	791.732
18¾	276.117	24⅞	485.978	32	804.249
18⅞	279.811	25	490.875	32¼	816.865
19	283.529	25⅛	495.796	32½	829.578
19⅛	287.272	25¼	500.741	32¾	842.390
19¼	291.039	25⅜	505.711	33	855.30
19⅜	294.831	25½	510.706	33¼	868.30
19½	298.648	25⅝	515.725	33½	881.41
19⅝	302.489	25¾	520.769	33¾	894.61
19¾	306.355	25⅞	525.837	34	907.92
19⅞	310.245	26	530.930	34¼	921.32
20	314.160	26⅛	536.047	34½	934.82
20⅛	318.099	26¼	541.189	34¾	948.41
20¼	322.063	26⅜	546.356	35	962.11

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

TABLE IV.—Continued.

Diam in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.	Diam. in Inches.	Area in Square Inches.
35¼	975.90	46½	1698.23	65½	3369.56
35½	989.80	46¾	1716.54	66	3421.20
35¾	1003.78	47	1734.94	66½	3473.23
36	1017.87	47¼	1753.45	67	3525.62
36¼	1032.06	47½	1772.05	67½	3578.47
36½	1046.35	47¾	1790.76	68	3631.68
36¾	1060.73	48	1809.56	68½	3685.29
37	1075.21	48¼	1828.46	69	3739.28
37¼	1089.79	48½	1847.45	69½	3793.67
37½	1104.46	48¾	1866.55	70	3848.46
37¾	1119.24	49	1885.74	70½	3903.63
38	1134.11	49¼	1905.03	71	3959.20
38¼	1149.08	49½	1924.42	71½	4015.16
38½	1164.15	49¾	1943.91	72	4071.51
38¾	1179.32	50	1963.50	72½	4128.25
39	1194.50	50½	2002.96	73	4185.39
39¼	1209.95	51	2042.82	73½	4242.92
39½	1225.42	51½	2083.07	74	4300.85
39¾	1240.08	52	2123.72	74½	4359.16
40	1256.60	52½	2164.75	75	4417.87
40¼	1272.39	53	2206.18	76	4536.47
40½	1288.25	53½	2248.01	77	4656.63
40¾	1304.20	54	2290.22	78	4778.37
41	1320.25	54½	2332.83	79	4901.68
41¼	1336.40	55	2375.83	80	5026.56
41½	1352.65	55½	2419.22	81	5153.00
41¾	1369.00	56	2463.01	82	5201.02
42	1385.44	56½	2507.19	83	5410.62
42¼	1401.98	57	2551.76	84	5541.78
42½	1418.62	57½	2596.72	85	5674.51
42¾	1435.56	58	2642.08	86	5808.81
43	1452.20	58½	2687.83	87	5944.69
43¼	1469.13	59	2733.97	88	6082.13
43½	1486.17	59½	2780.51	89	6221.15
43¾	1503.30	60	2827.44	90	6361.74
44	1520.53	60½	2874.76	91	6503.89
44¼	1537.86	61	2922.47	92	6647.62
44½	1555.28	61½	2970.57	93	6792.92
44¾	1572.81	62	3019.07	94	6939.79
45	1590.43	62½	3069.96	95	7088.23
45¼	1608.15	63	3117.25	96	7238.24
45½	1625.97	63½	3166.92	97	7389.80
45¾	1643.89	64	3216.99	98	7542.96
46	1661.90	64½	3267.46	99	7697.68
46¼	1680.01	65	3318.31	100	7854.00

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

ENGINE ECONOMY. — There are obvious losses in engineering practice, which render the actual amount of steam used per horse-power in excess of that calculated from the diagram. These losses are by *condensation* and *leakage*. There are two kinds of condensation: first, the radiation of heat from the engine-cylinder and steam-pipes into the external atmosphere; second, internal condensation, which is so great in amount that the gains expected from a high expansion of steam have been neutralized and limited, so that in one cylinder the best results are obtained with from only 3 to 5 expansions.

The loss due to internal condensation is a large one, and is due to the fact that the moist interior walls of the cylinder are cooled off during the period of expansion and exhaust, so that each fresh charge of steam entering the cylinder has to surrender a part of itself to the duty of warming up the cylinder to make it habitable for the remainder. The amount surrendered depends upon the amount of surface to be warmed, and the degree to which it has previously been made cold.

LONG *versus* SHORT STROKES.—The lost steam with early cut-offs, has amounted to over 40 per cent. of steam used. It bears a much larger proportion in engines of short than of long stroke, because the cold surface presented before cut-off takes place is much greater in short than in long stroke engines. Examine one end of two cylinders performing the same work, one 12″ diam. x 12″ stroke, and one 12″ diam. x 24″ stroke, piston moving at the rate of 600 feet per minute, giving 300 revolutions to the first, and 150 revolutions to the second case in each minute. Cut-off ¼ in each case.

The cold surfaces presented are:

	Short Stroke. sq. in.	Long Stroke. sq. in.
Surface of cylinder-heads	113	113
Surface of piston-heads	113	113
Surface of steam-ports, etc.	113 about	113 about
Surface of cylinder-walls up to ¼ cut-off	113	226
Total surface at one end of cylinder, per stroke, in sq. inches	452	565
" in sq. feet	3.1	3.9

Sq. feet exposed per minute by both ends of cylinder
$= 2 \times 3.1 \times 300 = 1{,}830$ square feet on short stroke.
$2 \times 3.9 \times 150 = 1{,}170$ square feet on long stroke.

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In this instance, if the cylinders are cooled down alike, the short stroke will condense about 60 per cent. more than the long stroke. There is no doubt that the short stroke engine, because of its higher speed, will not cool off as much as the long stroke; but experience has shown that this advantage is slight, compared with the great disproportionate amount of cooling surface.

To get good results on this score, long stroke engines at good piston speeds are desirable.

LEAKAGE is an insidious evil; its amount is seldom realized, perhaps not even suspected. There is no better service an engineer can render an employer than in keeping piston and valves tight. To ascertain if an engine leaks, test first the steam valves; to do this close both steam-ports by valve or valves and turn on steam, then observe through cylinder cocks or indicator holes if there be any leakage. To test exhaust-valves, close same, admit steam to cylinder with piston blocked in position, and examine the escape-pipe. To test piston, place on center with cylinder-head off and admit steam behind it. Valves can usually be made tight by scraping—unless they are piston or balanced valves. The best way to scrape valves is to take a little off at a time, scraping down the high places, which are worn bright by usage. If valves are red-leaded and scraped cold, the action of heat is such as to throw them out of shape when hot, and the bearing on seat may be far different when hot than cold.

Pistons should be made tight by good packing rings, or re-boring the cylinder if necessary.

The losses due to condensation and leakage combined are sometimes one-half of the steam used.

Mr. Geo. H. Barrus, of Boston, has prepared a table showing cylinder condensation for long stroke Corliss engines; if the steam used per indicated horse-power per hour, calculated from diagram at cut-off, is less than the percentage of feed water used per horse-power per hour, given by column 2, Table V., there is fair ground for assuming leakage.

EXAMPLE.—A diagram from a Corliss engine, cut-off ¼ stroke, gives at cut-off, 23.4 pounds of steam used per indicated horse-power per hour. The feed water, by measurement, is 36 pounds per indicated horse-power per hour. Percentage of steam accounted for $= \frac{23.4}{36} = 64$ per cent. By table this should be 76 per cent. Inference—a leak! Had feed water consumption been $\frac{23.4}{.76} = 31$ pounds per indicated horse-power per hour, it would have been evidence of no leakage.

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TABLE V.

PERCENTAGES OF LOSS BY CYLINDER CONDENSATION TAKEN AT CUT-OFF.

Compiled by Geo. H. Barrus, M. E.

Percentage of stroke completed at cut-off.	Percentage of Feed-Water Consumption accounted for by the indicator diagram.	Percentage of Feed-Water Consumption due to Cylinder Condensation.
5	58	42
10	66	34
15	71	29
20	74	26
30	78	22
40	82	18
50	86	14

COMPOUND ENGINES.

COMPOUND ENGINES are devices by which high grades o expansion, and consequently high pressure of steam, can b successfully used; and the evils of leakage also can be reduced

By expanding steam partially in one, then in a second, an perhaps a third cylinder, the internal condensation is kep small; for usually late cut-offs are employed, the advantag of which is that the surfaces of the cylinders presented fo re-heating to the fresh charges of steam are small, compare with the volume of steam used; and the difference i temperature between fresh and exhausted steam is slight.

Leakage of steam with compound engines is not so seriou a matter, the steam that leaks through one cylinder is caugh by the second instead of being thrown away. As the cut offs are frequently late, the slide-valve can be employed which is the tightest of all valves.

THE EFFECT OF LOAD UPON ECONOMY OF STEAM ENGINES.

A large engine working with an extremely light load i wasteful of fuel and steam; it is worse than a small busines with a large staff of expensive officers, for it has more tha the fixed charges always with it. A small engine overloade may be an annoyance and care, but if in good condition i may be economical in fuel.

ECONOMICAL PERFORMANCE OF STEAM ENGINES.

The following statements have been prepared as the result of experience as to the conditions, for each type of engine, from which best results can be obtained.

Non-Condensing Engines.

Slide-Valve Engine: 75 to 80 pounds of boiler pressure; stroke, long; mean effective pressure, 33 to 38 pounds per square inch; 25 to 100 horse-power; cut-off, $\frac{5}{8}$ stroke; about **40** pounds steam per indicated horse-power per hour. When valves and piston are tight this has been reduced to **33** pounds of dry steam per indicated horse-power per hour by careful test.

Automatic High-Speed Engines with single valves: 75 to 80 pounds of boiler pressure; stroke, about equal to piston diameter; m. e. p., 40 pounds per square inch; 50 to 150 horse-power; cut-off, $\frac{1}{4}$ stroke; about **40** pounds steam per indicated horse-power per hour. When valves and pistons are tight this has been reduced to **32** pounds of dry steam per indicated horse-power per hour. Valves difficult to keep tight.

Automatic High-Speed Engines with double valves: 75 to 80 pounds of boiler pressure; stroke $1\frac{1}{2}$ to 2 times piston diameter; m. e. p., 40 pounds per square inch; 50 to 150 horse-power; cut-off, $\frac{1}{4}$ stroke; about **35** pounds steam per indicated horse-power per hour. When valves and piston are tight this has been reduced to **30** pounds of dry steam per indicated horse-power per hour by careful test.

Automatic Cut-off Engines, of the Corliss type: stroke, 2 to 3 times diameter of piston; 75 to 90 pounds boiler pressure; m. e. p., 40 pounds per square inch; under 200 horse-power; cut-off, $\frac{1}{5}$ to $\frac{1}{4}$ stroke, **29-30** pounds steam per indicated horse-power per hour; over 200 horse-power, **27** pounds steam per indicated horse-power per hour; when valves and piston are tight, this has been reduced to **23⅛** pounds of dry steam per indicated horse-power per hour, by careful test.

Compound Engines; high speed, automatic cut-off, short stroke; 110 to 120 pounds boiler pressure; m. e. p., 25 to 27 pounds per square inch; 6 expansions, 100 to 250 horse-power; **27** pounds steam per indicated horse-power per hour.

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Condensing Engines.

Automatic Cut-off Engines, of the Corliss type; stroke, 2 to 3 times piston diameter; 70 to 80 pounds boiler pressure; m. e. p., 40 pounds per square inch; over 200 horse-power; cut-off, $\frac{1}{5}$ stroke, about 19-20 pounds steam per indicated horse-power per hour.

Compound Engines; high speed, automatic cut-off; short stroke; 110 to 120 pounds boiler pressure: m. e. p., 27 to 30 pounds per square inch: 9 expansions; 200 to 500 horse-power; 17 to 19 pounds steam per indicated horse-power per hour.

Compound Automatic Cut-off Engines, of the Corliss type; stroke on high pressure cylinder, 2 to 3 times piston diameter; 135 to 110 pounds boiler pressure; m. e. p., 24 to 14 pounds per square inch; over 400 horse-power; 16 to 20 expansions; 14 to 17 pounds of steam per indicated horse-power per hour. One or two special cases, 13½ pounds of steam per indicated horse-power per hour has been obtained.

ENGINE PROPORTIONS.

RESSURES ON WEARING SURFACES.

Main bearings.—140 to 160 pounds per square inch of area, obtained by multiplying length by diameter of journal.

Crank-pins.—1,000 to 1,200 pounds per square inch of area, obtained by multiplying length by diameter of pin.

Cross-head pins.—1,200 to 1,600 pounds per square inch of area, obtained by multiplying length by diameter of pin.

Cross-head surface.—35 to 40 pounds per square inch of area.

Non-condensing engines are usually designed for 100 pounds pressure per square inch of piston.

SIZES OF ENGINE PARTS, IN RELATION TO PISTON.

	Diameter of Piston.
Main-shaft, diameter	0.42 to 0.5
Main-bearing, length	0.85 to 1.0
Crank-pin, diameter	0.22 to 0.27
Crank-pin, length	0.25 to 0.3
Cross-head pin, diameter	0.18 to 0.2
Cross-head pin, length	0.25 to 0.3
Piston rod, diameter	0.14 to 0.17
Area steam ports:	Area of Piston.
Slide-valve engine	0.08 to 0.09
High-speed automatic engine	0.10 to 0.12
Corliss engine	0.07 to 0.8
Area exhaust ports:	
Slide-valve engine	0.15 to 0.2
High-speed automatic engine	0.18 to 0.22
Corliss engine	0.1 to 0.12

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Diameter of steam pipes:

Slide-valve engine, ¼ dia. piston to ¼ dia. piston + ½″.

Automatic high-speed engine, ⅓ diameter of piston.

Corliss engine, $\frac{3}{10}$ diameter of piston.

Diameter of exhaust pipes:

Slide-valve engine, ⅓ diameter of piston.

Automatic high-speed engine, ⅜ diameter of piston.

Corliss engine, ⅓ to ⅜ diameter of piston.

Clearance spaces:	Displacement of piston in one stroke.
Slide-valve engine	0.06 to 0.08
Auto. high-speed engine, single valve	0.08 to 0.15
Auto. high-speed engine, double valve	0.03 to 0.05
Auto. cut-off engine, Corliss type, long stroke	0.02 to 0.04

Weights of engines, per rated horse-power:

Slide-valve engine	125 to 135 pounds.
Auto. high-speed engine	90 to 120 pounds.
Corliss engine	220 to 250 pounds.

Fly-wheels, weight per rated horse-power:

Slide-valve engine	33 pounds.
Auto. high-speed engine (According to size and speed.)	25 to 33 pounds.
Corliss engine (According to size and speed.)	80 to 120 pounds.

RULES FOR FLY-WHEEL WEIGHTS, SINGLE CYLINDER ENGINES.

Let d = diameter of cylinder in inches.

S = stroke of cylinder in inches.

D = diameter of fly-wheel in feet.

R = revolutions per minute.

W = weight of fly-wheel in pounds.

For slide-valve engines, ordinary duty,	$W = \frac{350{,}000\ d^2\ S}{D^2\ R^2}$
For slide-valve engines, electric lighting,	$W = \frac{700{,}000\ d^2\ S}{D^2\ R^2}$
For automatic high-speed engines,	$W = \frac{1{,}000{,}000\ d^2\ S}{D^2\ R^2}$
For Corliss engines, ordinary duty,	$W = \frac{700{,}000\ d^2\ S}{D^2\ R^2}$
For Corliss engines, electric lighting,	$W = \frac{1{,}000{,}000\ d^2\ S}{D^2\ R^2}$

BOILERS.

The large majority of boilers now in use, are of the horizontal tubular type. These vary in size from 36″ dia. by 8 feet long to 72″ dia. by 20 feet long. The length depends upon

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the diameter of tubes used. The tube length should be 48 times the diameter for best results. Where soft coal is used a 4-inch tube gives the most satisfaction, as it does not rapidly clog up with soot and ashes. Therefore a boiler much shorter than 16 feet has not high efficiency in fuel.

No. of Tubes.—Tubes should not be placed too near together; 4-inch tubes should be arranged with at least 1″ spaces, and at least 2½ inch space between tubes and shell of boiler. When possible the middle vertical space between tubes should be 2 to 3 inches in width. Tubes should always be arranged in vertical rows to permit ease in cleaning.

TABLE VI.

Good Proportions for Tubular Boilers

Dia. Shell.	Length Shell.	No. of 4″ Tubes.	Horse Power.	Heating Surface.	Grate Area.	Chimney.	
						Diam.	Height.
inch.	feet.			sq. feet.	sq. feet.	inch.	feet.
48	16	28	50	600	16	24	60
54	16	32	60	769	18	27	60
60	16	46	75	917	25	30	60
66	16	54	90	1100	27½	33	60
72	16	64	100	1290	30	36	60

The tubes as given above permit a man-head in front, tube sheet below tubes.

Table VI. gives a very satisfactory arrangement of tubes, and also good proportions for grate and chimney. These boilers can be used without steam domes, as the water level is low enough for a dry pipe. In practice a greater number of tubes than here given are employed to secure slightly greater horse-power in a given shell; it is doubtful if this is not obtained by a loss in efficiency.

Dry Pipes perforated on the upper surface are liable to draw water that climbs up on the inside shell, and is sucked into the holes on top. The best arrangement consists of a pipe into which are screwed ½″ nipples about 1½″ long in two horizontal rows, one on each side of pipe. No water can be drawn into these openings unless boiler is nearly full. The area through nipples should be in excess of area of steam outlet. The ends of dry-pipe should be closed; there should be a hole in the bottom surface for water to drain out.

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Boilers from 36″ to 72″ diameter and 16, 18, and even 20 feet long have been built with the bottom sheet in one piece.

The points in favor of this method of construction are:

1st. No cross-seams in the fire.

2nd. There is no excuse for dirt to be left on bottom of boiler, as if there is any impediment to cleaning scraper, the operator knows it is dirt, not a rivet head.

3rd. Boiler is of more uniform strength; and consequently better adapted for expansion and contraction.

The objections:

1st. Difficult to patch.

2nd. Long sheets are difficult to bend.

3rd. Side-seams in large shells are exposed to fire.

Experience so far seems to indicate that boilers constructed in this manner are as durable and satisfactory as when built of circumferential rings.

All boilers are now made of steel, the quality of which has steadily improved from year to year. A good flange steel, that will possess between 55,000 and 60,000 pounds tensile strength and will turn down double cold without fracture, is a safe material.

The thickness of shells should not usually exceed ⅜″, or be less than ¼″; if greater than ⅜″ the metal is liable to burn, especially if circulation is poor; if less than ¼″, the metal is too thin, and liable to waste by corrosion. The usual thicknesses for different diameters is given by Table.

TABLE VII.

THICKNESS OF BOILER SHELLS AND HEADS FOR 100 POUNDS OF STEAM PRESSURE.

Diam. of Shell . .	36	40	42	46	48	52	54	60	66	72
Thickness of Shell .	¼	¼	¼	9/32	9/32	3/10	5/16	5/16	11/32	⅜
Thickness of Heads	⅜	⅜	⅜	7/16	½	½	½	½	½	½

A good rule for thickness of steel boiler shells is as follows:

Multiply the working pressure in pounds per square inch by diameter of shell in inches, divide by 19,200; the answer is thickness of shell in inches.

EXAMPLE.—Shell, 48 inches diameter; working pressure, 150 pounds.

What is thickness? $\frac{48 \times 150}{19,200} = \frac{3}{8}''$

Boiler Fittings usually consist of a safety-valve, a steam-gauge, a combination water column with water gauge and three gauge-cocks, the feed-valve, check-valve, and blow-off valve. The stop-valve is usually an extra fitting, dependent upon piping arrangements.

SAFETY-VALVES.—The following table gives the proper diameter and area of one common lever safety-valve in good order, for different hourly consumption of good coal, in boilers having a proper margin of strength above their working pressure.

TABLE IX.

Coal per hour.	Diameter of Valve in good order.	Area Opening Covered by Valve.
pound.	inches.	square inches.
17	1	.78
30	1 1/4	1.22
40	1 1/2	1.76
70	2	3.14
109	2 1/2	4.9
157	3	7.06
213	3 1/2	9.62
300	4	12.56

Valves, so constructed that they lift by reaction so high as to afford a clear opening around the valve equal to one-half the area of the opening covered by the valve, are safe for double the amount of coal indicated in column 1, if the boiler is otherwise adapted to the conditions.

Valves of the common type above 4 inches diameter are objectionable, and boilers consuming more than 300 pounds of coal per hour should have two smaller safety-valves with a combined area equal in square inches to the pounds of coal per hour, multiplied by 0.45.

Dividing the number of pounds of coal per hour by 5 will give roughly the horse-power of the boiler for this purpose.

EXAMPLE.—What size common lever safety-valve should be used with a 50 horse-power boiler? $5 \times 50 = 250$ pounds coal per hour; by table a 3½″ valve. If pop-valve, acting by reaction, is employed, $\frac{5 \times 50}{2} = 125$, or a 2½″ valve.

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The COMBINATION WATER COLUMN should always be connected to boiler by 1″ pipes, so that there is no danger from stoppage. The joints of piping should be made with crosses, the open ends stopped with brass plugs—by this means they can be cleaned out. There should also be a blow-off cock below column, through which mud can be blown out.

Remember! If the pipe is stopped up there is no way of determining the height of water in boiler, as the gauge cocks are screwed into column, and not into boiler.

The FEED-VALVE should be placed between the CHECK-VALVE and boiler. If the check-valve is disabled the feed-valve can be closed and check-valve taken apart and examined. If two or more boilers are connected to one main feed-pipe there should be a valve each side of check-valve.

The BLOW-OFF VALVE should be a good strong brass cock, or an asbestos packed cock. The pipe between blow-off cock and boiler should be large, as it is very apt to fill with sediment, scale, or mud.

Sizes for blow-off cock, pipe, feed, and check valves for different size boilers.

TABLE X.

SIZES OF BOILER FITTINGS.

Horse-power . .	30	35	40	45	50	55	60	70	75	85	100	125
	in.	in.	in.	in.	in.	in	in.	in.	in.	in.	inch.	inch.
Blow-off pipe .	1½	1½	1½	1½	1½	2	2	2	2	2½	2½	2½
Blow-off cock .	1¼	1¼	1¼	1¼	1¼	1½	1½	1½	1½	2	2	2
Feed-valve and check-valve .	1	1	1	1	1	1¼	1¼	1¼	1¼	1½	1½	1½

BOILER SETTINGS.

The brick-work about a boiler should be thick to prevent loss by radiation—a 21-inch wall should be used if possible. All flues and surfaces exposed to action of heat should be lined with best fire-brick.

It is not a good plan to convey gases back over top of boiler, unless there be space enough for a man to enter and clean off soot.

The distance from grate bars to lower portion of boiler shell should not be less than 24 inches; 26 and 28 inches are not too great, and in large shells 30 inches can be employed.

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The bridge-wall should curve to conform to shape of boiler shell. Ten (10) inches makes a good space between wall and shell.

Back of bridge-wall the surface should be paved with hard brick, the surface dipping down to a depth at rear end of boiler of about 18 to 24 inches according to size of shell.

The distance between back tube sheet and back wall should be 18 inches for a 48-inch shell; 24 inches for a 72-inch shell.

Boiler walls will crack, and no form of construction seems to entirely prevent this. Walls with air spaces are as liable as those without, with the danger of leaking more air when they do crack.

The best method to hold boiler walls together is with "buck-staves" or "buck-bars." The best form is railway iron with ends mashed down under a hammer, to allow for drilling holes for tie-rod. Most builders do not supply "buck-staves" unless specially ordered.

The cheapest form of fire-front is the so-called "half-arch," which does not cover any more of the front of the furnace than is absolutely decent. On small boilers it is employed as a support. For a good job a "full flush front" should be used with damper-plate and damper.

Boilers, now-a-days, are not set in batteries, all to work together as a unit. They are and should be set, so that each boiler is independent of the others in the battery. In this way any one can be shut down for cleaning or for repairs. This arrangement does away with the old-fashioned steam and mud-drums, which connected the boilers of a battery together. Do not buy either a mud-drum or steam-drum, they are a source of trouble, danger, and expense.

The trade usually includes with the boiler front, the grate-bars, a bearing-bar to support same, a soot or ash-door with frame, a back arch-plate or supporting bars, and a boiler stand for small boilers only.

HORSE-POWER OF BOILERS.

The standard of horse-power for boilers, the unit of which —30 pounds of water per hour, evaporated at 70 pounds pressure from 100° temperature of feed water—was selected by the judges at the Centennial Exhibition, at Philadelphia, in 1876, and has received the sanction of many engineers. In practice, however, this unit is far too small and represents

closely what good non-condensing automatic cut-off engines require in steam per horse-power per hour. If 48 or 50 pounds of water evaporated per hour were taken instead of 30 it would more nearly hit the usual idea of boiler horse-power. Many a purchaser has been caught because he has not been posted as to these facts. In reality the only test of boiler capacity is the amount of water evaporated per hour into steam from and at certain feed-water temperatures and steam pressures. The capacity also is influenced by the economy which is desired. If a boiler is pushed to its top capacity it will operate under poor economical conditions. If worked at too low a rate the conditions are not satisfactory. The best conditions for economy seem to be when each square foot of heating surface evaporates from 3 to 4 pounds of water into steam each hour, and from 16 to 20 pounds of soft coal are burned for each square foot of grate surface.

HEATING SURFACE.—With tubular boilers, 15 square feet of heating surface is often assumed to develop one horse-power. With small tubes closely spaced this is necessary, but with the proper number of large tubes amply spaced, 12 square feet of heating surface will accomplish nearly as good results. In the former case, with good coal and good draught, 3⅓ lbs. of water can be evaporated per sq. ft. per hour, or 50 pounds for each 15 square feet. In the latter case, other things equal, 4 pounds of water per square foot of heating surface can be evaporated per square foot per hour, or 48 pounds for each 12 square feet. The superior circulation of water in the latter case, presenting a greater number of particles to each unit of heating surface, is the secret of the superiority.

BOILER PERFORMANCE.—The amount of heat a pound of carbon will furnish by its combustion is about 14,500 thermal units. In the best boilers about 10,875 of these units can be utilized in evaporating water into steam; if the water is heated to enter boiler at 212°, and is simply converted into steam at that temperature we obtain a performance of $\frac{10,885}{965} = 11.2$ pounds of water evavorated by a pound of coal. (965 is the latent heat of steam at 212°. See table III.)

If the water is fed into boiler at 100° temperature

and evaporated at 75 pounds, the condition for standard horse-power, the boiler would evaporate $\frac{10,875}{1,108} = 9.8$ pounds of water per pounds of coal. (See example, page 5.)

Coal has a certain amount of impurity, which we will call non-combustible—frequently as much as 15 per cent. With this proportion 1 pound of coal as above would only evaporate 85 per cent. of 9.8 pounds equal to 8.3 pounds water per pound coal.

In view then, of this performance for the best class of boilers, a duty of 7½ pounds of water per pound of coal under ordinary conditions is not bad practice. A good feed water heater will help matters by putting water into boiler at or nearly at 212°. The following table shows state of case:

TABLE XI

POUNDS OF WATER EVAPORATED PER POUNDS OF COAL UNDER DIFFERENT CONDITIONS.

	Temp. of feed water 212°. Evap. at 212°	Temp. of feed water 100°. Evap. at 320°, corresponding to 75 lbs. boiler pres. by gauge.	Temp. of feed water 200.° Evap. at 331° corresponding to 90 lbs. boiler pres. by gauge.
	pounds.	pounds.	pounds.
Theoretical conditions, 1 lb. of pure carbon . .	15	13	14.3
1 lb. of pure combustible under best boilers . . .	11.2	9¾	10.7
Best coal under best boilers 8 % non-combustible . .	10.3	9	9.9
Fair coal under best boilers 15 % non-combustible .	9.8	8¼	9.1
Fair coal under fair boilers 15 % non-combustible .	8.3	7	7.7
Poor coal under fair boilers 30 % non-combustible .	6.8	5¾	6.3

All above horizontal lines are theoretical conditions.

The best possible practice is thus shown to be 9.9 pounds water to 1 pound coal, actual conditions.

Thus, the boasts of evaporating 9 and 10 pounds of water per pound of coal is usually absurd, and is only obtainable with the best fuel, best boilers, and best methods of heating feed water.

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Feed water heaters are a most important feature of a good steam plant. First; by utilizing the heat of the exhaust steam from the engine or waste gases in chimney, the feed

TABLE XII.

SHOWING GAIN IN USE OF FEED WATER HEATER. PERCENTAGE OF HEAT REQUIRED TO HEAT WATER FOR DIFFERENT FEED AND BOILING TEMPERATURES, AS COMPARED WITH A FEED AND BOILING TEMP. OF 212°.

Boiling Point Fahr.	Initial Temperature of feed water.											
	32°	50°	68°	86°	104°	122°	140°	158°	176°	194°	212°
212	1.19	1.17	1.15	1.13	1.11	1.10	1.08	1.06	1.04	1.02	1.00
230	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02	1.01
248	1.20	1.18	1.16	1.14	1.13	1.11	1.09	1.07	1.05	1.03	1.01
266	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06	1.04	1.02
284	1.21	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04	1.02
302	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07	1.05	1.03
320	1.22	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.05	1.03
338	1.23	1.21	1.19	1.17	1.15	1.14	1.12	1.10	1.08	1.06	1.04
356	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	1.04
374	1.24	1.22	1.20	1.18	1.17	1.15	1.13	1.11	1.09	1.07	1.05
392	1.24	1.23	1.21	1.19	1.17	1.15	1.13	1.11	1.09	1.07	1.06
410	1.25	1.23	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06
428	1.25	1.24	1.22	1.20	1.18	1.16	1.14	1.12	1.11	1.09	1.07

water may be heated to about 210° F. with ease, before entering boilers; by this means saving fuel, and increasing capacity of boiler. Second; by heating the water the boilers are protected from serious and unequal strain, as the

difference of temperature between incoming water and outgoing steam may be kept about 110°, (210° to 320°.) Third; every heater must necessarily be a water purifier, as the mud and lime are eliminated, to some degree at least, before the water reaches the boiler by heat.

There are two distinct types of Heaters, in which heat is derived from exhaust steam. These are known as closed and open heaters. Each has its advantages and disadvantages.

The CLOSED HEATER is constructed so that the water is forced under pressure through tubes or chambers surrounded by the exhaust steam, the heat being transmitted through the walls of the tubes and chambers.

The OPEN HEATER is a vessel in which the feed water comes into direct contact with the exhaust steam, by spraying or intermingling. The heated water is pumped hot into the boiler.

The CLOSED HEATER has the advantage of permitting the water to pass through the pump cold, and in that state is easily handled. To pump hot water from an open heater requires special care in piping and packing the feed-pump.

The closed heater, being a purifier (if any lime is present in water, a portion is bound to be precipitated by heat), should be cleaned, a job about as difficult as cleaning a boiler; or blown out, which is never a satisfactory method. In the precipitation of lime by heat, carbonic acid gas is set free, and the chemists say that this gas in a nascent state (just being born) attacks iron and brass. Whatever the cause, experience has demonstrated that ordinary wrought iron, steel and brass, corrode under this action, and it is only a matter of time before serious leaks appear. Therefore, closed heaters are not durable.

The OPEN HEATER, being usually a large chamber, is accessible for cleaning out, and if made with ordinary care will last a long time. A leak in it is not a serious matter, while a leak in the closed heater means a waste of hot water into the exhaust pipe.

The open heater has at times been the cause of serious mishaps. In it the steam and water mix; with any stoppage in exit of feed water, there is danger of flooding the cylinder of the steam engine through exhaust pipe, causing a wreck. The more modern forms of these heaters and the experience obtained in their use have reduced this difficulty to a minimum. The remaining

difficulty—pumping hot water—will be considered under "Pumps." In view of these facts, the open heater is now being more generally used than formerly.

PUMPS.

Duplex pumps are largely used for all purposes, on account of their durability and easy action. For boiler feeders they have some points of inferiority, especially for hot water; but taken all in all, they are the most satisfactory pump made.

DUPLEX PUMPS FOR BOILER FEEDING.—The speed should not be over 50 single strokes of one plunger per minute, or on long strokes 100 feet piston speed per minute.

TABLE XIII.

SIZES FOR BOILER FEEDING.

Diam. of Steam Cylinder.	Diam. of Water Cylinder.	Stroke.	Horse-Power Boilers.	Steam Pipe.	Exh'st Pipe.	Suct'n Pipe.	Disch. Pipe.
3½	2¼	4	30- 40	⅜	½	1	¾
4½	2¾	4	80-100	½	¾	2	1¼
5½	3½	5	140-160	¾	1¼	2½	1½

Use size larger suction pipe, if long suction is required. A usual allowance for boiler feeding is to deliver 1 cubic foot or 7½ gallons of water per horse-power per hour.

PUMPING HOT WATER.—To do this successfully certain conditions must be observed; the suction and discharge-valves must be located above the pump-barrel, so that it is constantly submerged; the pump-plunger should be long and large, so that clearance spaces at each end of pump are as small as possible. Linen packing put in loose should be used on pump-plunger; rubber will decompose under heat.

A vapor pipe should be attached to the suction pipes between heater and pump; it consists of a vertical pipe, higher than the heater, open at the top; this permits the air and vapor entangled in hot water to escape, and prevents them passing through pump. A valve can be placed in this pipe. It is also a good plan to screw into pump-chamber, just below the discharge-valve plate, a nipple, a check-valve opening outward, a nipple and angle-valve; through this the vapor

liberated in pump can any time be discharged by simply opening the angle-valve.

An air chamber is not needed on a boiler feeder when placed near the boiler, with direct pipe connections.

In ordering a pump for hot water, be sure the dealer or manufacturer understands that fact.

HORSE-POWER OF GEARS.

To determine the horse-power which any gear-wheel will transmit, four facts are required to be known:

1st. The kind of wheel, whether spur, bevel, spur mortise or bevel mortise.

2d. The pitch.

3d. The face.

4th. The velocity of pitch circle in feet per second.

Generally the fourth fact is not known. It can be found if the pitch diameter of the wheel in inches, and the number of revolutions per minute are given, for it can be obtained from them by the following rule:

RULE.—Given, the pitch diameter, in inches, and the number of revolutions per minute; to find, the velocity of pitch line in feet per second.

First, multiply the pitch diameter (in inches) by the number of revolutions per minute.

Second, divide the product thus found by 230.

The quotient is the velocity required.

EXAMPLE.—What is the velocity of the pitch diameter of a gear-wheel in feet per second, the pitch diameter = 43 inches, the revolutions per minute = 125?

$$43 \times 125 \div 230 = 23.4 \text{ feet per second.}$$

Table XIV. shows the greatest horse-power which different kinds of gears of 1-inch pitch and 1-inch face will safely transmit at various pitch-line velocities. To find the greatest horse-power which any other pitch and face will safely transmit, the following rule can be used:

RULE.—Given, the pitch (in inches), face (in inches), velocity of pitch circle (in feet per second), and kind of gear; to find, the greatest horse-power that can be safely transmitted.

First; find the horse-power in Table XIV., which the given kind of wheel with 1-inch pitch and 1-inch face will transmit at the given velocity.

Second; multiply the pitch by the face.

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Third; multiply the horse-power found, by the product of pitch by face.

The final product is the horse-power required.

EXAMPLE.—What is the greatest horse-power that a bevel-wheel, 43″ pitch diameter, 2″ pitch, 6″ face, and 125 revolutions per minute will safely transmit?

From previous example we have found the pitch-line velocity to be 23.4 feet per second, which is nearest to a velocity of 24 feet per second in table XIV.

First, the horse-power which a bevel-wheel of 1″ pitch and 1″ face will transmit is (from table), at this velocity, 4.931.

Second, the product of pitch by face is $2 \times 6 = 12$.

Third, $12 \times 4.931 = 59.17$ horse-power. Answer.

Whenever it is desirable to know about the average horse-power that any wheel will transmit, $\frac{2}{3}$ or $\frac{1}{2}$ of the results obtained by the rule above should be taken.

TABLE XIV.

TABLE SHOWING THE HORSE-POWER WHICH DIFFERENT KINDS OF GEAR WHEELS OF ONE INCH PITCH AND ONE INCH FACE WILL TRANSMIT AT VARIOUS VELOCITIES OF PITCH CIRCLE.

1	2	3	4	5
Velocity of Pitch Circle in Feet per Second.	Spur Wheels.	Spur Mortise Wheels.	Bevel Wheels.	Bevel Mortise Wheels.
2	1.338	.647	.938	.647
3	1.756	.971	1.227	.856
6	2.782	1.76	1.76	1.363
12	4.43	3.1	3.1	2.16
18	5.793	4.058	4.058	2.847
24	7.052	4.931	4.931	3.447
30	8.182	5.727	5.727	4.036
36	9.163	6.314	6.414	4.516
42	10.156	7.102	7.102	4.963
48	10.083	7.680	7.680	5.411

NOTE.—When velocities are given, which are between those in table, the horse-power can be found by interpolation.

Thus the horse-power for spur-wheels at 14 feet velocity is found as follows:

$14 - 12 = 2$
$18 - 12 = 6$ | $5.793 - 4.43 = 1.363$.

Then 2-6 of $1.363 = .454$.

and $.454 + 4.43 = 4.884$ horse-power.

TABLE XV.— SHAFTING.

HORSE-POWER TRANSMITTED BY VARIOUS SHAFTS, AT 100 REVOLUTIONS PER MINUTE UNDER VARIOUS CONDITIONS.

1	2	3	4
Diameter of Shaft.	Line Shafts.	Shaft as a Prime Mover.	Shafts Under Slight Bending Strain.
$\frac{15}{16}''$	.7	.4	1.3
$1\frac{3}{16}''$	1.3	.7	2.6
$1\frac{7}{16}''$	2.4	1.2	4.7
$1\frac{11}{16}''$	3.8	1.9	7.6
$1\frac{15}{16}''$	5.8	2.9	11.5
$2\frac{3}{16}''$	8.3	4.2	16.6
$2\frac{7}{16}''$	11.5	5.8	23.
$2\frac{11}{16}''$	15.5	7.8	31.
$2\frac{15}{16}''$	20.	10.	40.
$3\frac{3}{16}''$	26.	13.	51.
$3\frac{7}{16}''$	33.	17.	65.
$3\frac{11}{16}''$	40.	20.	80.
$3\frac{15}{11}''$	49.	25.	97.
$4\frac{7}{16}''$	70.	35.	139.
$4\frac{15}{16}''$	96.	48.	192.
$5\frac{7}{16}''$	126.	64.	256.
$5\frac{15}{16}''$	167.	84.	334.
$6\frac{15}{16}''$	266.	133.	532.
$7\frac{15}{8}''$	399.	200.	797.
$8\frac{15}{16}''$	570.	285.	1139.
$9\frac{15}{16}''$	783.	392.	1566.

This table states the horse-power that various sizes of shafts will safely transmit at 100 revolutions per minute under various conditions.

PRIME MOVERS are those shafts in which the variation above and below the average horse-power transmitted is great, also where the transverse strain due to belts or heavy pulleys is large, such as jack-shafts, crank-shafts, etc.

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To find the horse-power for any shaft for any speed different from 100, multiply suitable number for size of shaft given in table by the actual speed of shaft and divide by 100.

EXAMPLE.—What horse-power will a line-shaft 2 15-16″ diameter transmit, speed 160 revolutions per minute?

$20 \times 160 \div 100 = 32$ horse-power.

TABLE XVI.

SHOWING DIAMETER OF SHAFTS FOR SINGLE BELTS OF VARIOUS WIDTHS, THE DISTANCE BETWEEN HANGERS IMAGINED 1 FOOT.

1	2
Diameter of Shaft.	Width of Belt in Inches.
$1\frac{5}{16}''$	5″
$1\frac{3}{16}''$	11″
$1\frac{7}{16}''$	19″
$1\frac{1}{16}''$	31″
$1\frac{5}{16}''$	47″
$2\frac{3}{16}''$	68″
$2\frac{7}{16}''$	94″
$2\frac{1}{16}''$	126″
$2\frac{5}{16}''$	165″
$3\frac{3}{16}''$	211″
$3\frac{7}{16}''$	264″
$3\frac{1}{16}''$	326″
$3\frac{5}{16}''$	398″
$4\frac{3}{16}''$	569″
$4\frac{5}{16}''$	785″
$5\frac{7}{16}''$	1048″
$5\frac{5}{16}''$	1365″

This table may be used to find the largest size belt any shaft will carry safely, when the kind of belt, the position of the pulley on shaft, and the distance between bearings are known.

RULE.—Given: single belt, pulley quarter-way from

bearing, diameter of shaft, and distance between bearings.

Divide number in column 2 (opposite to size of shaft) by span in feet.

EXAMPLE.—Single belt, pulley quarter-way from bearing, 2 15-16″ shaft, 8 ft. span.

165″ ÷ 8 = 21″ width of single belt.

If pulley is situated half-way between bearings, divide by twice the span.

EXAMPLE,—Single belt, pulley half-way between bearings, 2 15-16″ shaft, 8 ft. span.

2 × 8 = 16. 165 ÷ 16 = 10½″ width of belt.

RULE.—Given: double belt, pulley quarter-way from bearing, diameter of shaft, and distance between bearings.

Divide number in column 2 (opposite to size of shaft) by twice the span in feet.

EXAMPLE.—Double belt, pulley quarter-way from bearing, 3 7-16″ shaft, span 10 ft.

10 × 2 = 20. 264 ÷ 20 = 13″ width of belt.

If pulley is situated half-way between bearings, divide by four times the span.

EXAMPLE.—Double belt, pulley half-way between bearings, 3 7-16″ shaft, span 10 ft.

10 × 4 = 40. 264 ÷ 40 = 6½″ width of belt.

To allow for weight of pulley, divide weight by 100 and the quotient is equivalent to a single belt of that width: that is, a pulley weighing 1,200 pounds would strain the shaft as much as a 12″ single belt.

BELTING.

To ascertain horse-power which belts will transmit, multiply width of belt by diameter of pulley (in inches), by revolutions of pulley (per minute), by number in table (corresponding to the pull the belt can exert per inch of width.)

EXAMPLE.—10″ single horizontal belt, 36″ pulley, 200 revolutions, pull taken at 50 pounds.

10″ × 36″ × 200 × 0.0004 = 28.8 horse-power.

The pulls which belts 1″ wide will transmit are as follows:

Single horizontal belts (pulleys nearly same diameter),	50 lbs.
Double " " " " " "	100 "
Single vertical " " " " " "	40 "
Double " " " " " "	60 "
Single belts (large to very small pulleys)	10 "
Double " " " "	15 "
Quarter twist, single belts	25 "
" double "	40 "

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Pull exerted by Belt 1" Wide. In lbs.	HORSE-POWER. Pulley 1" Diameter, 1 Revolution per Minute—Belt 1" Wide.
10	.00008
15	.00012
20	.00016
25	.0002
30	.00024
35	.00028
40	.00032
45	.00036
50	.0004
55	.00044
60	.00048
65	.00052
70	.00056
75	.0006
80	.00064
85	.00068
90	.00072
95	.00076
100	.00080

SWEET'S VALVE DIAGRAM.

To find outside and inside lap of valve for different cut-offs and compressions: (See figure, page 36.)

Draw a circle whose diameter equals travel of valve.

Draw diameter B A and continue to A_1 so that the length A A_1 bears the same ratio to X A, as the length of connecting rod does to length of engine crank.

Draw small circle E with a diameter equal to lead.

Lay off A C, so that ratio of A C to A B = cut-off, in parts of the stroke.

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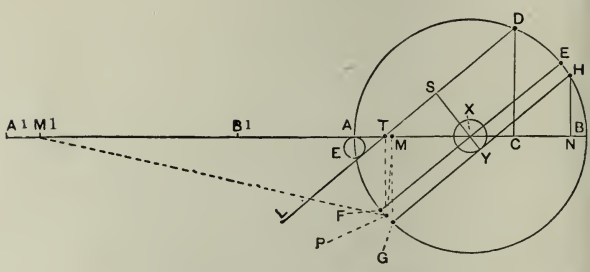
Erect perpendicular C D. Draw D L tangent to E. Draw X S perpendicular to D L. X S is then outside lap of valve.

To find release and compression. If there is no inside lap, draw F E through X parallel to D L. F and E will be position of crank for release and compression. If there is an inside lap, draw a circle about X, in which radius X Y equals inside lap. Draw H G tangent to this circle and parallel to D L; then H and G are crank position for release and compression. Draw H N and M G, then A N is piston position at release and A M piston position at compression, A B being considered stroke of engine.



SWEET'S VALVE DIAGRAM.

To make compression alike on each stroke it is necessary to increase the inside lap on crank end of valve, and to decrease by the same amount the inside lap on back end of valve. To determine this amount, through M with a radius M M_1 = A A_1, draw arc M P, from P draw P T perpendicular to A B, then T M is the amount to be added to inside lap on crank end, and to be deducted from inside lap on back end of valve, inside lap being X Y.

CENTRIFUGAL FORCE.—The centrifugal force of a body depends upon its weight, W in lbs.; distance, R, in feet it is from the center of rotation; and the number of revolutions, N, it makes about that center each minute, and equals

$$\frac{W\ R\ N^2}{2933}$$

Multiply the weight in pounds by radius in feet, by square of number of revolutions, and divide by 2933 = centrifugal force in pounds.

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

SIZES OF CHIMNEYS WITH APPROPRIATE HORSE-POWER BOILERS.

Babcock & Wilcox Co.'s Catalogue.

Diam. in Inches.	Height of Chimneys. Commercial Horse-Power.										Effective Area, square feet.	Actual Area, square feet.	Side of Square of Approximate Area, Inches.
	50 ft.	60 ft.	70 ft.	80 ft.	90 ft.	100 ft.	110 ft.	125 ft.	150 ft.	175 ft.	200 ft.			
18	23	25	27									0.97	1.77	16
21	35	38	41									1.47	2.41	19
24	49	54	58	62								2.08	3.14	22
27	65	72	78	83								2.78	3.98	24
30	84	92	100	107	113							3.58	4.91	27
33		115	125	133	141							4.47	5.94	30
36		141	152	163	173	182						5.47	7.07	32
39			183	196	208	219						6.57	8.30	35
42			216	231	245	258	271					7.76	9.62	38
48				311	330	348	365	389				10.44	12.57	43
54				363	427	449	472	503	551			13.51	15.90	48
60				505	539	565	593	632	692	.748		16.98	19.64	54
66					658	694	728	775	849	918	981	20.83	23.76	59
72					792	835	876	934	1023	1015	1181	25.08	28.27	64
78						995	1038	1107	1212	1310	1400	29.73	33.18	70
84						1163	1214	1294	1418	1531	1637	34.76	38.48	75
90						1344	1415	1496	1639	1770	1893	40.19	44.18	80
96						1537	1616	1720	1875	2027	2167	46.10	50.27	86

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

HANDY ROUGH RULES FOR POWER.

$$\text{Horse-power of engine} = \frac{P\ L\ A\ N}{16500}$$

P = Average pressure throughout stroke, in pounds, per square inch, (see tables.)
L = Length of stroke, in feet.
A = Area of piston, in square inches.
N = Number of revolutions per minute.

Multiply them all together and divide their product by 16500.

$$\text{Horse-power of shaft} = \frac{D3 \times N}{100} \text{ nearly.}$$

Cube the diameter in inches, multiply same by number of revolutions per minute, and divide product by 100. A shaft one inch in diameter, at 100 revolutions per minute, will usually transmit one horse-power.

$$\text{Horse-power of belting} = \frac{W\ D\ N}{2000} \text{ nearly.}$$

Multiply width of belt in inches by diameter of pulley in inches, by number of revolutions per minute, and divide product by 2000. A belt one inch wide, moving over a pulley 10 inches in diameter at 100 revolutions, will give ½ horse-power.

$$\text{Horse-power of spur gearing} = \frac{P\ F\ D\ N}{600} \text{ nearly.}$$

Multiply pitch of wheel, face and diameter, all in inches, together, by the number of revolutions per minute, and divide by 600.

TABLE XIX.

VELOCITIES.

I. Velocities of Belts, Ropes and Gearing.	Vel. in feet per minute.
Minimum velocity of belts	200
Average velocity of belts	2000
Maximum velocity of belts	5000–6000
Minimum velocity of wire ropes, transmitting power continuously,	1000
Average velocity of ditto	3600
Maximum velocity of ditto	6000
Average velocity of pitch circles of gear wheels, transmitting power continuously	750–1500
Maximum velocity of ditto	2750
II. Velocities of Cutting Edges for Metals.	
Average velocities when turning, planing, or shaping articles of steel or gray cast iron	10–20

Mean velocity when turning, planing, or shaping articles of white cast iron	5
Mean velocity when turning chilled rolls	3
Average velocities in planing wide grooves, as key-seating, etc.	10–12
Average velocities in planing, turning, or shaping articles of wrought iron	15–24
Average velocities in planing, turning, or shaping articles of brass and bronze	40–100
Average velocities of shears for plates	36
Average velocities of shears in cutting bars of iron and steel	12
Average velocity of periphery of plate bending rolls	4
Mean velocity of periphery of iron, steel, copper, etc.—articles of large diameter, when being polished with emery and oil,	750
Mean velocity of periphery of grindstones, for polishing iron and steel	1800
Best velocity of periphery of emery wheels for polishing	5280

III. Velocities of Cutting Edges for Wood-Working Machines.

Velocity of periphery of circular saws, sawing with grain	9000
Ditto, sawing across grain	7200
Velocity of lumber being sawed, (average)	240–360
Velocity of band saws	10000
Average velocity of periphery of bits for boring	24
Maximum velocity of periphery of bits for boring	80
Velocity of periphery of large work in lathes	2400
Velocity of periphery of small work in lathes	900
Velocity of edges of planer bits	7200

IV. Velocities of Pistons, Etc.

Mean velocity of pump pistons	60
Maximum velocity of pump pistons	150
Ordinary velocity of pistons for steam engines	400
Average velocity of pistons for high speed steam engines	600
Maximum velocity of pistons of locomotives	1200
Velocity of pistons in engines of Porter-Allen types	900
Maximum velocity of pistons in air-pumps	420

V. Velocities of Rims of Fly-Wheels.

For ordinary engines	2500
For rolling-mill engines	5280
Maximum theoretical velocity	8500

VI. Velocities for Flour and Oil Mills.

Velocities of crushing rolls at periphery	60-180
Maximum velocity of grain, flour, etc., elevators	165
Mean velocity for grain, flour, etc., elevators	100
Mean velocity of rollers for squeezing oil out of seeds	140
Average velocity for periphery of bolting reels for cleaning grain, etc	225
Ditto for cleaning flour	240

The H-S-G Slide-Valve Steam Engine is the Standard.

Maximum velocity at which light flour, etc., may be carried on linen bands, without being dispersed by air currents	50
Ditto, for heavy grains	54
Velocity of periphery of mill-stones	1333–2000
Average speed of conveyors	30

VII. Velocities of Water, Air, Gas and Steam.

Average velocity of water in pump	200
Maximum velocity of water in tubes	500
Best velocity of water in clay channels	30
Average velocity of rivers	240
Maximum velocity of air passing through fire grates of steam boilers	210
Velocity of air in moderate weather	780–1320
Best velocity of air in pipes of blowing engines	200
Maximum velocity of air through valves of blowing engines	250
Average velocity of gases in chimneys, with artificial draught,	100
Best velocity for illuminating gas in pipes	700
Velocity of steam under 1.5 atmospheres' pressure, entering freely into air	67500
Ditto, under 3 atmospheres' pressure	98400
Ditto, under 5 atmospheres' pressure	110580
Velocity of steam under 1 atmosphere pressure, entering freely into vacuum	114540

VIII. Shafting.

	Revs. per minute.
Main shafts for machine shops	125
Main shafts for wood-working machinery	250
Main shafts for cotton and woolen mills	300–400

SLIDE-VALVE ENGINES FOR ELECTRIC LIGHTING.

The Houston, Stanwood & Gamble slide-valve engine is especially well adapted for driving isolated dynamos, belted direct, when slow speeds are desired.

Upon special order these engines are supplied for this purpose with a special finish, governor one size larger than the standard, a heavy fly-wheel of large diameter and continuous sight-feed oilers throughout. So arranged, these engines are suitable for heavy work and continuous running.

Fittings Usually Furnished with Standard Slide-valve Engines by Houston, Stanwood & Gamble.

Governor, with Automatic Satety Stop.
Fly band wheel.
Stop valve.
Automatic Sight-feed Cylinder Lubricator.
2 Oil Cups for Connecting-rod.
2 Oil Cups for Slides.
Cylinder Cocks.
Drain Cock in Stop-valve.
1 Wrench for Piston-rod.
1 Double End Wrench, fitting Connecting-rod and Stuffing Boxes.

When specially ordered, the Foundation rods with nuts and washers, and the Governor belt is supplied.

TABLE XX.

HORSE-POWER OF HOUSTON, STANWOOD & GAMBLE'S SLIDE-VALVE STEAM ENGINES.

HORSE-POWER OF
7x12
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 Lbs.	75 Lbs.	90 Lbs.
200	14	16	18
225	16	18	**20**
250	17	**20**	**22**

HORSE-POWER OF
8x12
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 Lbs.	75 Lbs.	90 Lbs.
200	16	18	21
225	18	20	**23**
250	20	**22**	**26**

HORSE-POWER OF
8x14
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 Lbs.	75 Lbs.	90 Lbs.
175	18	21	24
200	20	24	**27**
225	23	**27**	**31**

HORSE-POWER O.
9x14
ENGINE AT

Rev. per Min.	Boiler Pressure 60 Lbs.	75 Lbs.	90 Lbs.
175	24	28	32
200	27	31	**35**
225	30	**35**	**40**

HORSE-POWER OF
10x14
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 Lbs.	75 Lbs.	90 Lbs.
175	27	30	34
200	31	35	**40**
225	35	**40**	**46**

HORSE-POWER OF
10x16
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 Lbs.	75 Lbs.	90 Lbs.
150	30	36	42
180	36	43	**50**
210	42	**50**	**56**

TABLE XX.—Continued.

HORSE-POWER OF HOUSTON, STANWOOD & GAMBLE'S SLIDE-VALVE STEAM ENGINES.

HORSE-POWER OF
11x18
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
150	35	42	50
180	42	50	**58**
210	50	**58**	**66**

HORSE-POWER OF
12x16
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
150	40	47	54
180	47	54	**63**
210	54	**63**	**72**

HORSE-POWER OF
12x18
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
140	40	48	56
170	48	57	**67**
200	56	**66**	**78**

HORSE-POWER OF
13x20
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
140	50	58	66
170	58	69	**79**
200	66	**80**	**92**

HORSE-POWER OF
14x20
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
130	60	72	84
150	67	84	**96**
170	75	**96**	**108**

HORSE-POWER OF
15x22
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
130	70	82	94
150	80	94	**107**
170	90	**106**	**120**

TABLE XX.—Continued.

HORSE-POWER OF HOUSTON, STANWOOD & GAMBLE'S SLIDE-VALVE STEAM ENGINES.

HORSE-POWER OF
16 x 22
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
130	80	93	106
150	92	106	**121**
170	104	**119**	**136**

HORSE-POWER OF
17 x 24
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
125	110	130	150
135	120	140	**162**
145	128	**150**	**175**

HORSE-POWER OF
18 x 26
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
120	125	146	168
130	135	160	**185**
140	145	**172**	**200**

HORSE-POWER OF
20 x 28
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
110	150	175	200
120	165	182	**220**
130	180	**210**	**240**

HORSE-POWER OF
22 x 30
ENGINE AT

Rev. per Min.	Boiler Pressure. 60 lbs.	75 lbs.	90 lbs.
100	170	200	226
110	187	220	**250**
120	204	**240**	**280**

Heavy Proportions.
Simple Design.
Generous Wearing Surfaces.
Breakdowns Reduced to A Minimum.

SIZES OF THE

H-S-G
TRADE MARK.

SLIDE-VALVE STEAM ENGINE.

RIGHT OR LEFT HAND.

SELF-CONTAINED OR WITH OUTBOARD BEARING SEPARATE.

DOUBLE AND WITH LINK MOTION.

STANDARD SLIDE-VALVE STEAM ENGINES.

NOMINAL HORSE-POWER	CYLINDER.		BAND FLY-WHEEL.			SIZE OF GOVERNOR AND STEAM PIPE.	SIZE OF EXHAUST PIPE	REV. PER MIN. TO GIVE NOMINAL H. P.	SHAFT.	
	Bore.	Stroke	Diam.	Face.	Weight in Rough.				Diam.	Length
	Inches	Inches	Inches	Inches	Lbs.	Inches	Inches		Inches	Inches
12	7	12	42	8	385	1½	3	200	3⅜	44
18	8	14	48	10	730	2	3	175	3⅞	48
24	9	14	54	12	940	2½	3½	175	3⅞	48
30	10	16	60	14	1170	2½	3½	150	4⅞	60
40	11	18	66	14	1450	3	4	150	4⅞	60
50	12	18	72	16	1800	3	4	150	5⅞	69
60	13	20	78	16	2000	3½	4½	140	5⅞	69
70	14	20	84	17	2475	3½	4½	130	5⅞	69
80	15	22	96	17	3000	4	5	130	6⅞	78
100	16	22	96	20	3600	4½	6	130	6⅞	78

SHORT-STROKE ENGINES.

16	8	12	42	8	385	2	3	200	3⅜	44
27	10	14	54	12	940	2½	3½	175	4⅜	48
45	12	17	66	14	1450	3	3	150	4⅞	60

HEAVY-DUTY ENGINES.

NOMINAL HORSE-POWER	CYLINDER.		BAND FLY-WHEEL.			SIZE OF GOVERNOR AND STEAM PIPE.	SIZE OF EXHAUST PIPE.	REV. PER MIN. TO GIVE NOMINAL H. P.	SHAFT.	
	Bore.	Stroke.	Dia.	Face.	Weight in Rough.				Dia.	Length
	Inches	Inches	Inches	Inches	Lbs.	Inches	Inches		Inches	Inches
100	15	22	108	20	4200	4	5	140	6⅞	78
110	16	22	108	20	4800	4½	6	130	6⅞	78
125	17	24	120	20	5600	4½	6	120	7⅞	90
150	18	26	120	25	6600	5	6	120	7⅞	90
175	20	28	144	25	8800	5	7	110	8⅞	96
200	22	30	144	35	11000	6	7	100	8⅞	96

THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.

The H-S-G TRADE MARK.

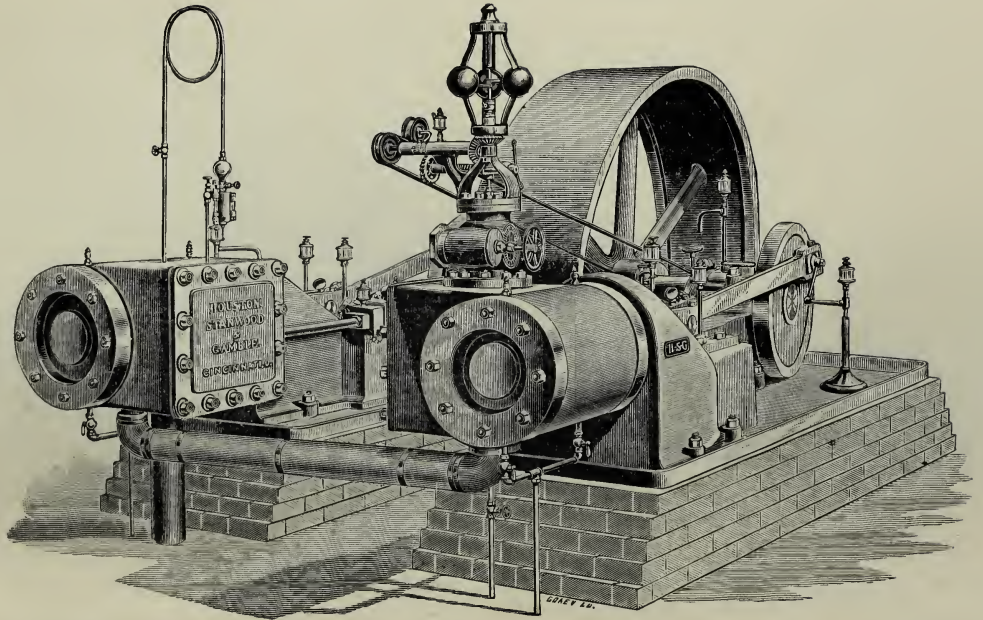
CROSS COMPOUND NON-CONDENSING SLIDE-VALVE ENGINE.

FOR BOILER PRESSURES 80 TO 120 LBS.

SIZES AND PROPORTIONS.

HORSE POWER (RATED) AT 100 LBS. BOILER PRESS'RE	REV. PER MIN'TE TO GIVE RATED HORSE POWER	NUMBER DESIGNATING SIZE OF ENGINE.	MAXIMUM BOILER PRESSURE ALOWD LBS.PR SQ. IN	WHEEL.				GOVERNOR & STEAM PIPE, SIZE. INCHES	EXHAUST PIPE. SIZE. INCHES
				DIAMETER. INCHES	FACE. INCHES	WEIGHT IN ROUGH. LBS.	BORE. INCHES		
25	215	13	110	66	9	1000	3⅞	2	3
30	210	16	120	72	9	1100	4⅞	2	3
33	225	16½	100	66	10	1250	3⅞	2½	3½
38	225	19	120	72	10	1400	4⅞	2½	3½
40	220	20	120	66	12	1600	4⅞	2½	4
43	200	23	120	72	12	1800	4⅞	2½	4
48	225	24	100	66	13	2000	4⅞	3	4
50	200	28	100	72	13	2200	4⅞	3	4
55	185	32	120	78	14	2400	5⅞	3	4
60	180	37	120	84	14	2400	5⅞	3	4½
65	175	42	120	90	15	2650	5⅞	3	4½
70	180	43	120	84	15	3000	5⅞	3½	4½
75	170	49	120	90	15	3000	5⅞	3½	4½
80	175	50	110	96	15	3300	5⅞	3½	5
85	166	56	110	96	15	3500	5⅞	3½	5
90	175	57	100	96	17	3800	5⅞	4	6
95	166	64	100	96	19	4400	5⅞	4	6
100	155	71	120	108	19	5000	6⅞	4	6
105	162	72	120	108	19	5500	6⅞	4	6
110	152	80	120	108	21	6600	6⅞	4	6
120	166	81	120	108	21	7700	6⅞	4½	7
125	154	90	120	120	21	8300	6⅞	4½	7
135	150	105	100	120	21	8800	6⅞	4½	7
140	135	116	120	120	25	10000	7⅞	4½	7
150	130	128	120	120	25	11000	7⅞	4½	7
170	130	147	120	120	27	12500	7⅞	5	8
180	125	160	120	144	27	13500	8⅞	5	8
200	120	190	120	144	29	14500	8⅞	6	8
225	120	206	120	156	29	16000	8⅞	6	8
275	120	259	110	156	31	17000	8⅞	6	9
300	120	279	120	168	33	18000	9⅞	6	9
340	110	344	120	168	35	19000	9⅞	7	10
360	110	368	110	180	37	20000	9⅞	7	10

TO FIND FULL INDICATED POWER OF ENGINE: MULTIPLY NUMBER WHICH DESIGNATES SIZE OF ENGINE, BY BOILER PRESSURE AND REVOLUTIONS PER MINUTE; DIVIDE PRODUCT BY 10,000; QUOTIENT=MAXIMUM INDICATED HORSE-POWER.



The New H-S-G Throttling Non-Condensing Cross Compound Steam Engine.

ADVANTAGES
OF THE
CROSS COMPOUND NON-CONDENSING SLIDE-VALVE STEAM ENGINE.

THE work is evenly divided between high and low pressure cylinders, with light and heavy loads.

GREAT steadiness of motion, as cranks are at right angles and fly-wheel does not have to carry load when cranks pass their centers.

CAN be underspeeded 50 per cent., on account of high efficiency of fly-wheel, so that large engines may be installed at low speeds for temporary light loads.

BREAK-DOWNS reduced to a minimum, on account of great simplicity of construction and strength of parts.

IN cases of repair to either high or low pressure engine, the one or the other engine may be operated under control of the governor with half load.

NO danger from water, as slide-valves on sides of cylinders give easy relief to water in cylinders.

MAY be operated continuously, as all parts can be lubricated when engine is in motion.

SPEED can be changed when the engine is in motion.

STEAM-JACKET on high pressure cylinder heats its exhaust steam as it passes on to the low pressure cylinder.

LOSSES through leakage are reduced to a minimum as the slide-valve is the tightest form of valve made.

SMALL FLUCTUATION of speed with sudden changes of load, as nearly entire inertia of fly-wheel is available to resist rapid external changes.

PISTONS are accessible and are easily removed, which is not the case with the Tandem Compounds.

STRAINS are nearly proportioned to loads, as initial pressures vary with the loads. With automatic cut-offs strains are as great with light as with heavy loads. On some compounds they are greatest with lightest loads.

MAXIMUM strains on all wearing parts are one-half of that of Tandem Compounds.

THESE engines have been operated without fly-wheels.

EACH engine is mounted upon a cast iron base or foundation plate which catches all oil and drip.

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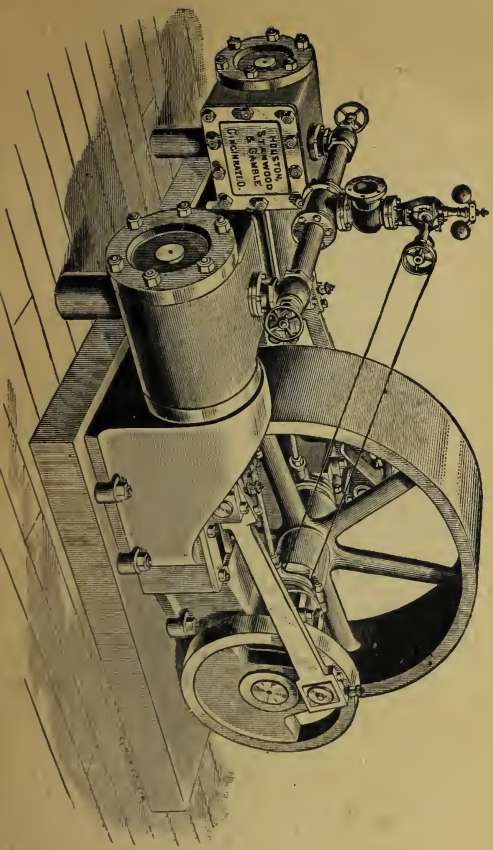
THE H-S-G SLIDE-VALVE STEAM ENGINE IS THE STANDARD.



The Houston, Stanwood & Gamble Double Engine.

HOUSTON, STANWOOD & GAMBLE,

SPECIALISTS,

MANUFACTURERS ONLY OF

SLIDE-VALVE STEAM ENGINES,

FACTORY: SOUTH END C. & O. RY. BRIDGE,

CINCINNATI, O.

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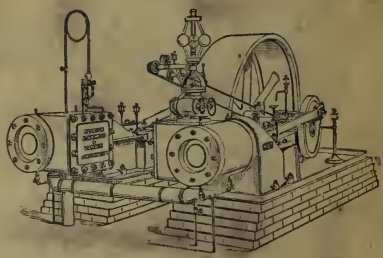
THE NEW H-S-G THROTTLING
NON-CONDENSING CROSS COMPOUND
STEAM ENGINE.



BUILT BY
HOUSTON, STANWOOD & GAMBLE,
CINCINNATI, OHIO.